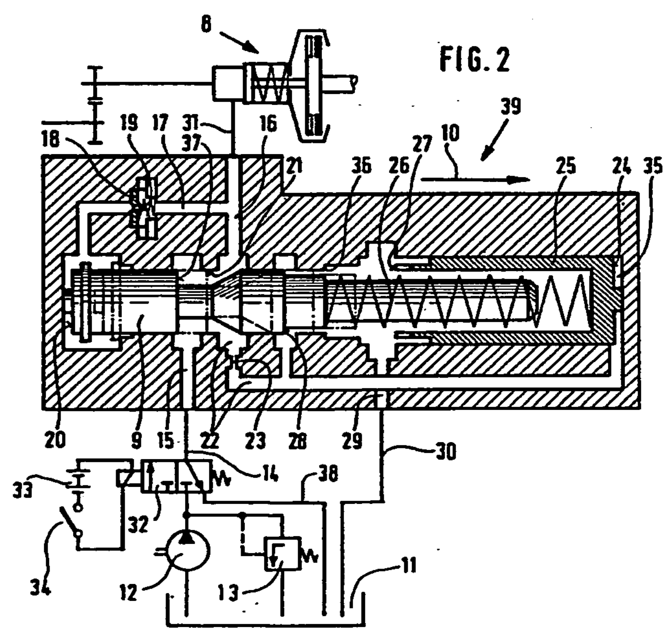
(12) UK Patent Application (19) GB (11) 2 106 610 A

(21) Application No 8222697
(22) Date of filing 6 Aug 1982
(30) Priority data
(31) 3131420
(32) 7 Aug 1981
(33) Fed Rep of Germany (DE)
(43) Application published
13 Apr 1983
(51) INT CL³
F16D 25/14 65/32
(52) Domestic classification
F2L 8B2U 8B3C 9A2 9BU
F2F GD GE
(56) Documents cited
GB 1208193
GB 1191429
(58) Field of search
F2D
F2F
F2L
(71) Applicants
Zahnradfabrik
Friedrichshafen AG
(FR Germany),
Friedrichshafen-on-the-Bodensee,
Federal Republic of
Germany
(72) Inventor
Franz Schele
(74) Agents
Venner Shipley and Co.,
368 City Road,
London EC1V 2QA

(54) **Pressure control valve for clutch or brake**

(57) A pressure control valve (39) is inserted in a fluid controlled system, in particular for the operation of fluid controlled clutches and brakes, in which increasing pressure is used for the actuation of the pressure controlled cylinder of the fluid controlled element (8), and in which the actuation of the cylinder takes place at a considerably increased speed with respect to conventional devices. For this purpose fluid is supplied to the pressure control valve (39) via a supply line (15) with the movable valve elements (9, 25) initially in positions as shown, but flow is then reduced by fluid acting on a control surface (20) of the element (9) which is supplied via a feedback control line (17) incorporating a flow-restricting diaphragm (18). Another control line (22) incorporating a flow-restricting orifice (23) supplies fluid to the control piston element (25) which acts on the element (9) via a spring (26). A fluid return line (29, 30) is also provided, by which the pressure in line (22) is relieved until a land (28) of the element (9) cuts off communication between those lines.

FIG. 2



GB 2 106 610 A

BEST AVAILABLE COPY

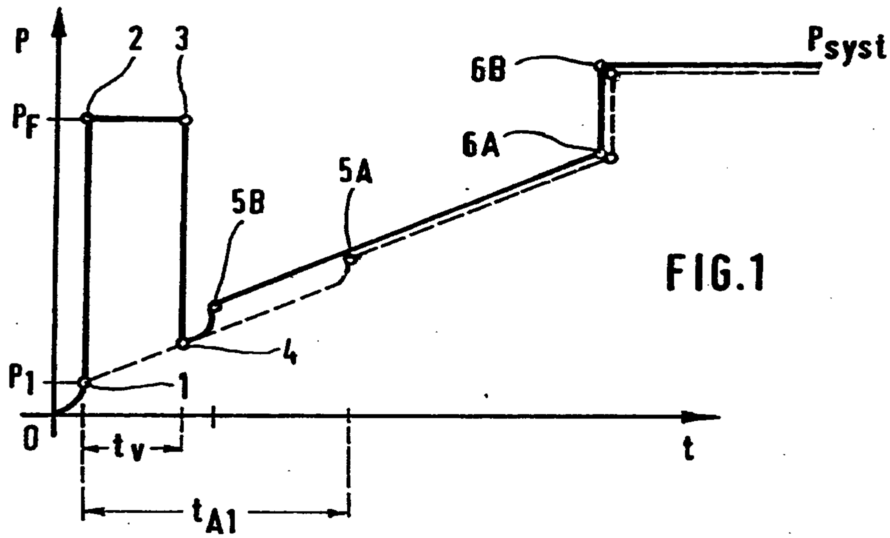

P
2
3
P_F
6B
P_{syst}
6A
5A
5B
4
P_1
1
0
t_v
t
t_{A1}

FIG.1

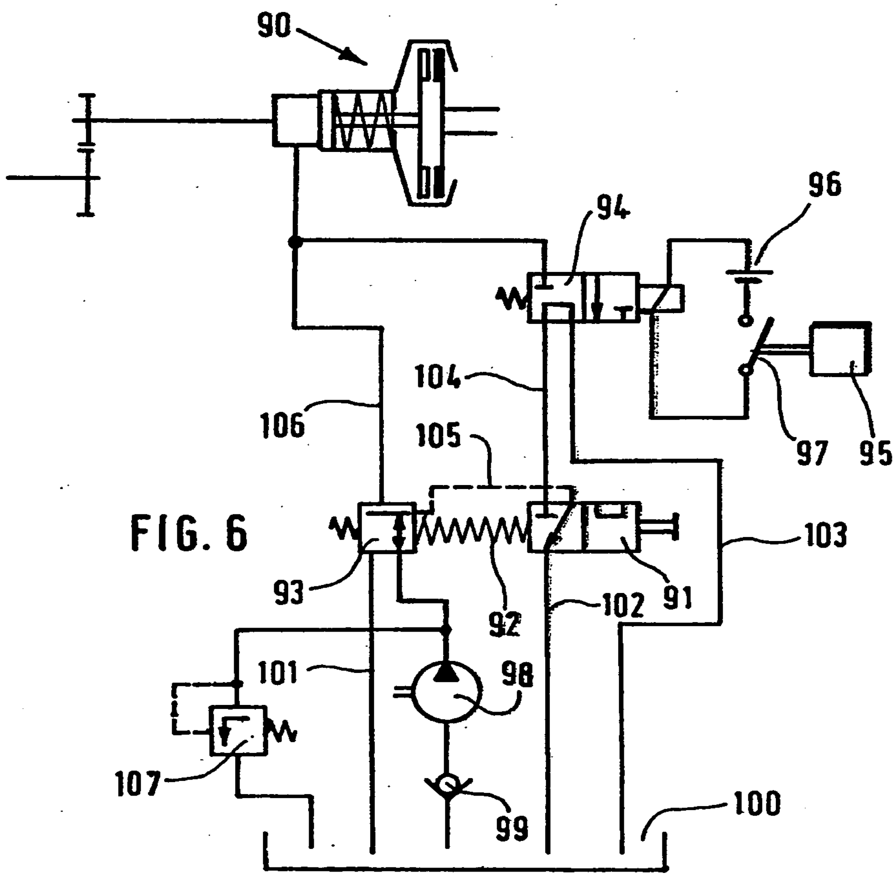

90
94
96
104
106
105
97
95
103
93
92
102
91
101
98
107
99
100

FIG. 6

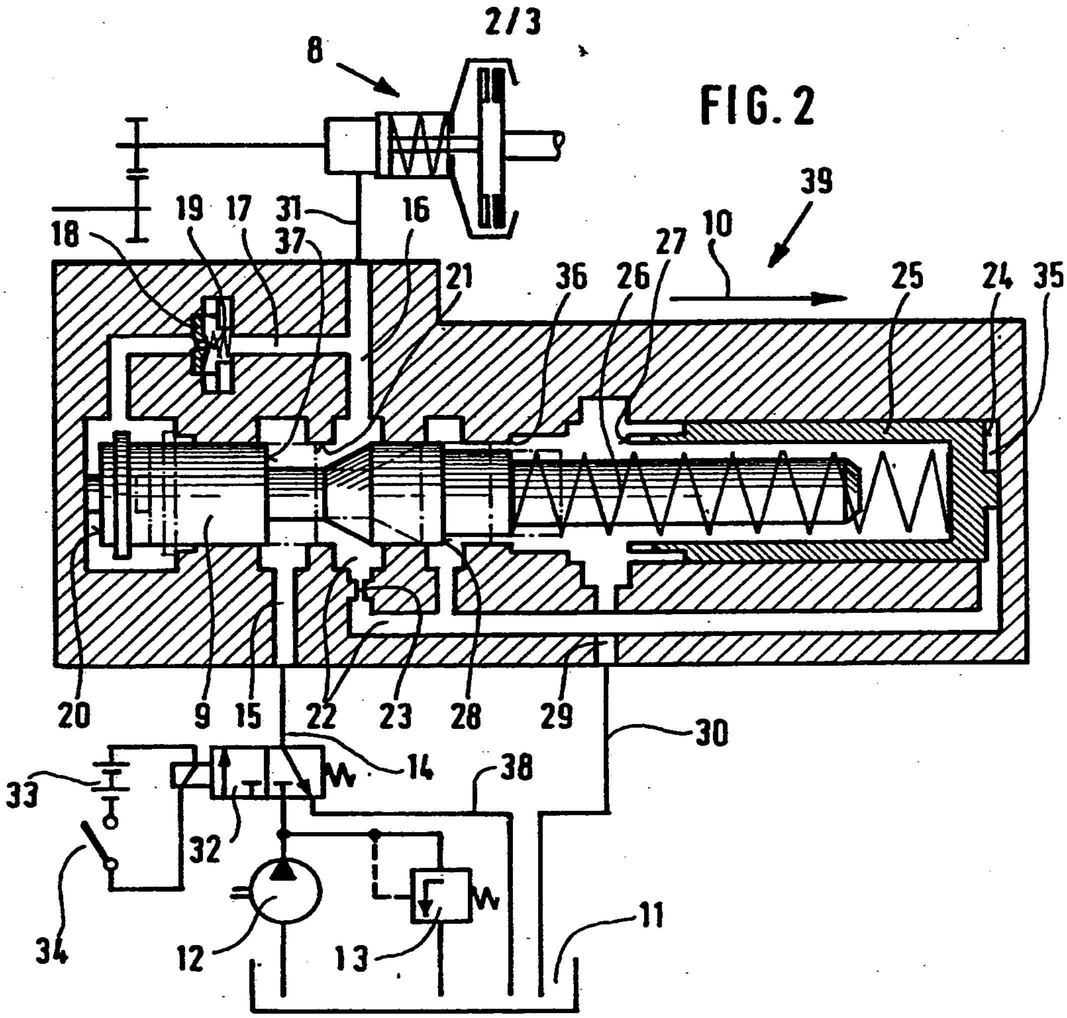

8
39
10
18
19
17
31
37
16
21
36
26
27
25
24
35
20
9
15
22
23
28
29
30
14
38
33
32
34
12
13
11

FIG. 2

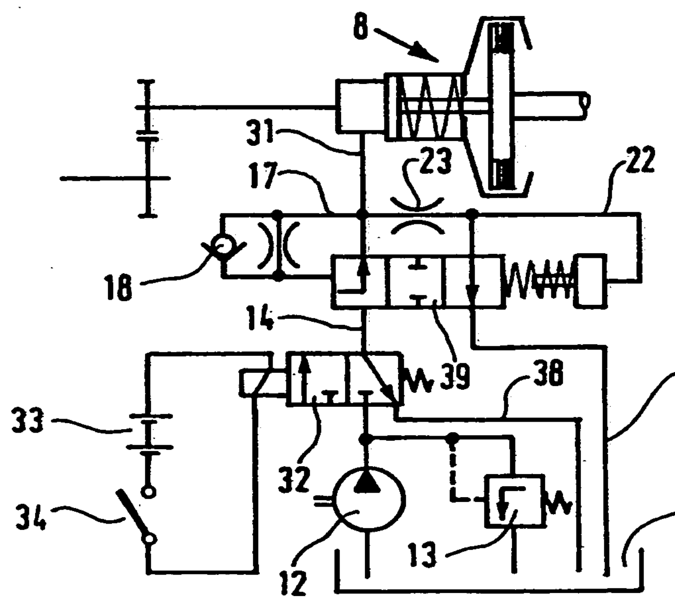

8
31
23
22
17
18
14
39
38
30
33
32
34
12
13
11

FIG.3

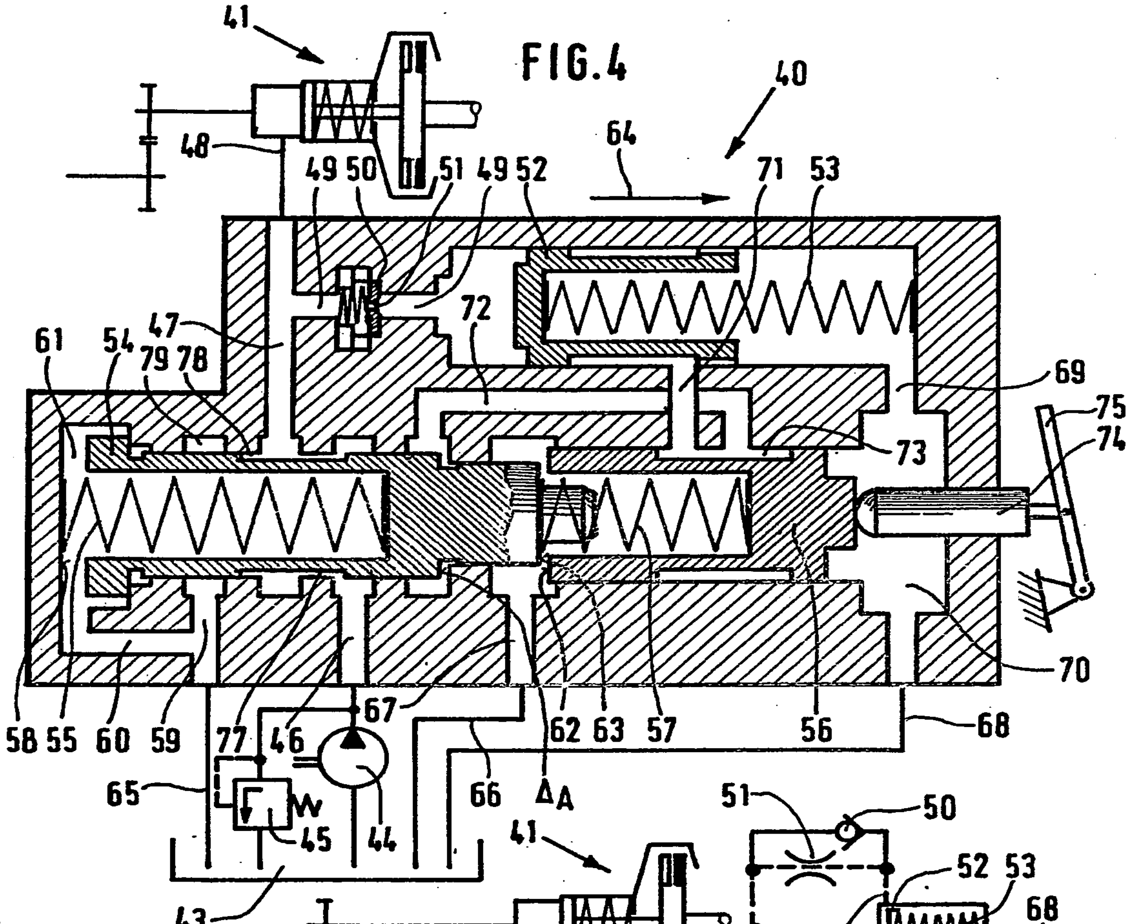


FIG.4

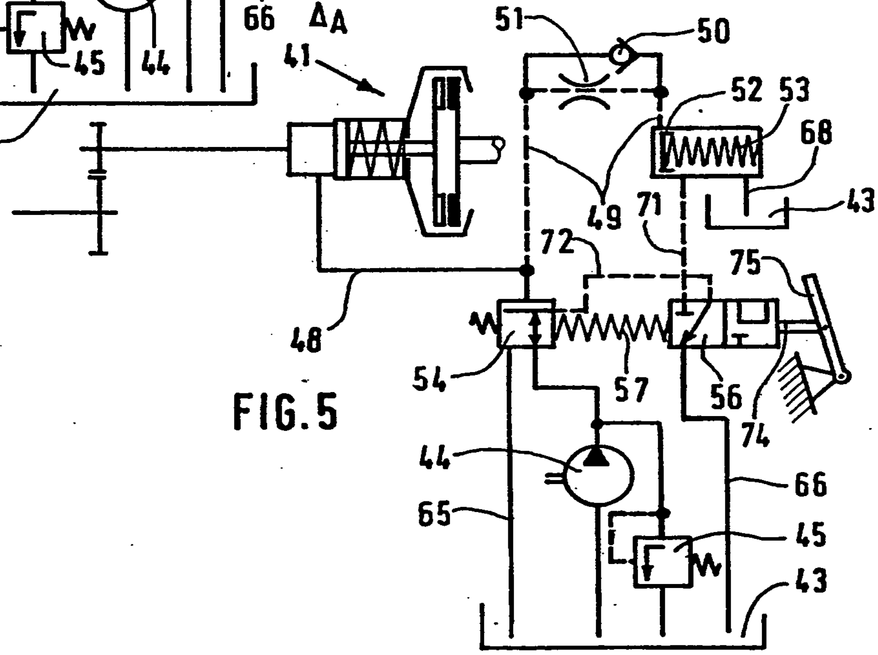


FIG.5

SPECIFICATION

Device for use in a fluid controlled system for rapid actuation of a cylinder

The invention relates to a device for use in a fluid controlled system for the rapid actuation of a pressure controlled cylinder, preferably acting as part of a clutch or a brake and having control valve connected in series with the cylinder.

In fluid operated systems, in which force is transmitted and controlled within a switching circuit by pressurized fluids, a pressure decrease takes place as a result of the pressure control valve used in each case during a first stage in which the fluid controlled elements must only be supplied with a certain amount of the pressure medium, during the period when a predetermined pressure is not yet required during this stage. The pressure decrease during this stage produces a low fluid flow speed and extends the first stage as can be seen from the fact that the fluid controlled elements have a delayed response.

The object of the present invention is to make the first stage, for filling purposes, therefore the time required for the empty stroke, as short as possible, in order to enable the response of, for example a clutch of a brake, i.e. an operating cylinder, to be as early as possible.

In accordance with the invention this object is solved by the features set out in the characterising part of claim 1.

With respect to known devices, the invention is advantageous in that the response time of fluid controlled elements may be considerably shortened. Consequently it is possible to rapidly fill a pressure cylinder, i.e. a clutch or brake, for the purpose of the actuation of the corresponding element and to enable actuation of the element after a short period.

Further advantageous embodiments are set out in the sub-claims and the following description.

The invention is exemplified with respect to embodiments thereof described with reference to the accompanying drawings, in which:–

Fig. 1 is a graph showing the course of a filling pressure as a function of time which may be achieved with the device of the invention.

Fig. 2 is a diagrammatic illustration of a device in accordance with the invention incorporating a pressure control valve,

Fig. 3 is a circuit diagram of the embodiment of Fig. 2,

Fig. 4 shows a further embodiment of pressure control valve having a retarding piston,

Fig. 5 is a circuit diagram of the embodiment of Fig. 4, and

Fig. 6 is a circuit diagram of a further embodiment incorporating a different timing element.

It is attempted with this invention rapidly to fill a fluid controlled element with fluid, i.e. to produce a high filling pressure P_F (Fig. 1) as rapidly as possible after opening of a valve connected in series with the fluid controlled element.

In the case of known systems, after opening of a series-connected valve, the sustem pressure located in front of a pressure control velve is reduced from the outset such that fluid starts to be supplied to the fluid controlled element to be actuated after reaching a pressure P_1 (Fig. 1). This reduced filling pressure is at point 1, and filling then proceeds according to the dashed line between points 4 and 5A. As a result of the low filling pressure and the low fluid flow speed resulting therefrom, the amount of fluid required for the empty stroke is only available to the fluid controlled element after a response time t_{A1}, so that the fluid controlled element is only in fact operative from the point 5A.

However, in the case of the invention, the pressure is not, in the first instance, decreased after opening of the valve. In contrast, after passing point 1, point 2 is directly reached, which corresponds to a filling pressure P_F which is not decreased. As a result of the high filling pressure P_F, the fluid flows very rapidly, so that within a short period the required fluid amount is supplied to the fluid controlled element for its empty stroke. Shortly before the amount of fluid required for the empty stroke is supplied to the fluid controlled element, the pressure decrease of the pressure control valve commences, so that after a delay period t_v, the fluid pressure decreases from point 3 (pressure P_F) to a point 4 (Fig. 1) from which the remaining fluid is supplied to the fluid controlled element at a reduced pressure. The fluid supply only continues up to the point of time shown by 5B, so that the fluid controlled element is in fact operative from point 5B.

After the amount of fluid required in each case for the empty stroke has been supplied to the fluid controlled element, the fluid becomes stationary in the supply lines leading to the fluid controlled element. In this respect a slight pressure increase occurs behind the pressure control valve (points 5A and 5B). After reaching points 5B and then 5A, the pressure in the fluid controlled element and in the supply line from the pressure control valve to the fluid controlled element increases up to point 6A. Subsequently a point 6B may be reached which corresponds to the system pressure P_{syst} if this is higher than the pressure at the point 6A.

The hypothetical ideal curve of Fig. 1 may be achieved using a device as shown in Figs. 2 and 3. A fluid controlled clutch 8 constituting the fluid controlled element in this embodiment is connected via a line 31 to the outlet of a pressure control valve 39. It would also be possible to connect a fluid controlled brake or a different element to the output of the fluid controlled valve 39.

A junction valve 32 which is electromagnetically actuated against a return spring and which receives current from a battery 33 on closing of a switch 34, is connected via a line 14 to a line 15 forming the input of the pressure control valve 39 (Figs. 2, 3). The junction valve 32 is connected to the delivery side of a pump 12 which is disposed in parallel with a pressure limiting valve 13.

This print takes account of replacement documents later filed to enable the application to comply with the formal requirements of the Patents Rules 1982.

A pressure control valve 39 (Fig. 2) comprises a piston means which is constituted by an operating piston 9 and a control piston 25. If there is no pressure at the pressure control valve 39, the operating piston 9 and the control piston 25 are located in the position shown in the drawing. In this condition the line 15 is connected via the line 14, the junction valve 32 and a further line 38, with the reservoir 11. In this respect, the line 15 of the pressure control valve 39 is in connection with line 16, so that the pressure chamber of a clutch 8 is pressureless.

If the switch 34 is closed, the junction valve is switched from its locked position and the pressure originating from the pump 12 is supplied to the pressure control valve 39 via line 15. It is then transmitted via the lines 16 and 31 to the pressure chamber of the cylinder of the clutch 8 and causes a fluid flow at this point so that the piston of the clutch is moved. The pressure prevailing in the line 16 is a function, on one hand, of the supply stream and, on the other hand, of the flow resistances in the lines and in the fluid controlled element, for example in the clutch 8. As a result of the prevailing pressure, a small amount of fluid is also simultaneously urged through lines 17 and 22, and the fluid passes through a diaphragm 18 or a throttle 23. The pressure produced in this way, on a control surface 20 of the operating piston 9, displaces said piston against the force of a compression spring 26 in the direction of the arrow 10 (Fig. 2). After the operating piston 9 has covered a short distance, a projection 28 separates the control line 22 from a return line 29. The fluid flowing through the throttle 23 bears on a control surface 24 of the control piston 25 in a chamber 35. The control piston is slowly displaced in the opposite direction to the arrow 10 under the compression of the compression spring 26. The operating piston 9 moves further in the direction of the arrow 10 until a control edge 37 of the free cross-section of a bore 21 and therefore the fluid stream is reduced to such an extent that the pressure in the line 16 and the pressure acting on the control surface 20 decreases until the force acting on the control surface 20 is equal in size to the instantaneous force of the compression spring 26. The pressure prevailing in the line 31 consequently drops to point 4 (Fig. 1). If the pressure prevailing in the line 16 and the pressure on the control surface 20 is greater than that corresponding to the force of the spring 26, the bore 21 is further closed and the fluid stream is further throttled or even completely discontinued. However if the pressure drops too far, the force of the spring 26 effect a further opening of the bore 21 as a result of which the fluid stream increases. The valve 26, therefore, has passing through it the exact amount of fluid required for maintaining the pressure correpsonding to the instantaneous spring tension.

As the control piston 25 of the pressure control valve 39 is displaced further in the opposite direction to the arrow 10, there is a constant increase in the force of the compression spring 26. The controlled pressure therefore also increases.

The diameter of a bore 19 in the diaphragm 18 is dimensioned such that the fluid flowing through the bore 19 only displaces the operating piston 9 in the direction of the arrow 10 slowly such that the flow of fluid through the bore 21 is only throttled shortly before the clutch 8 is closed. This makes it possible for the fluid passing through the supply line 15, the bore 21 and the lines 16, 31 and coming from the pump 12 to flow freely to the clutch 8 whilst its cylinder is filled and it performs the major part of its empty stroke. The pressure is not therefore reduced during the major portion of the filling stage. The diaphragm 18 inserted in the line 17 therefore causes a more rapid flow of the fluid through the supply line 15, the bore 21 and the lines 16, 31, whereas if the diaphragm 18 were not provided, the operating piston 9 would be displaced so far in the direction of the arrow 10 that it would almost close off the bore 21 and therefore the lines 16, 31 would only admit a pressure corresponding to quotients of the force of the compression spring 26 and the control surface 20 on the operating piston 9.

As the control piston 25 (Fig. 2) is displaced in the opposite direction to the arrow 10, the pressure prevailing in the lines 16, 31 increases in accordance with the distance 5B, 6A (Fig. 1), whose increase may be adjusted by the dimensioning of the throttle 23.

After the control piston 25 has travelled a certain distance in the direction opposite to that of the arrow 10, it contacts the operating piston 9 and displaces it in this direction until the control piston 25 has a free annular edge 27 adjacent to an edge 36 rigid with the housing. The bore 21 is thereby opened and there is a connection between the supply line 15 and the line 16 so that the same pressure prevails in both lines. In the diaphragm (Fig. 1) the distance 6A, 6B extends up to system pressure P_{syst}.

The pressure control valve 39 may be formed such that the path of a control piston 25 through the edge 36 acting as a stop is limited and the operating piston 9 is unable to contact the control piston 25. In this case the adjusting function of the pressure control valve 39 is not offset so that the pressure increase along the distance 6A, 6B (Figs. 1) is dispensed with. The compression spring 26 may also be embodied such that the adjusted pressure has already reached the value of the system pressure P_{syst} before the control piston 25 comes into contact with the operating piston 9.

If the switch 34 is opened (Fig. 2, 3), the valve 32 returns to its locked position and the pressure in the supply line 15 and in the lines 16, 31 decreases, the pressure control valve 39 being evacuated via a line 30. The pressure acting on the control surface 20 decreases and the operating piston 9 is displaced by the tensioned compression spring 26 in the opposite direction to the arrow 10. In this respect the diaphragm 18 opens and releases a greater flow crosssection so that the operating piston 9 may return rapidly to its initial position. The projection 28 also releases the bore which it has closed so that the chamber 35 is evacuated via the return lines 29, 30 and the control piston 25 also returns to its initial position as shown in Fig. 2 (Point 0 in Fig. 1).

Figs. 4 and 5 show a further embodiment. A pump 44 disposed in parallel with a pressure limiting valve 45 supplies from a reservoir 43 a fluid which is supplied to the supply line 46 forming an inlet of a pres-

sure control valve 40. A fluid controlled clutch 41 is connected via a line 48 to a line 47 leading to the outlet. An operating piston 54 and a control piston 56 are provided in the pressure control valve 40. On the operating piston 54 a compression spring 55 is supported against the housing of the pressure control valve 40, and a further compression spring 57 is tensioned between the operating piston 54 and the control piston 56. The pressure control valve 40 also comprises a retarding piston 52 against which a compression spring 53 is supported, the other end of this spring being adjacent to the housing. A diaphragm 50 with a bore 51 is inserted along the course of a control line 49 leading from the line 47 to the retarding piston 52.

When the pedal 75 is pressed, in order to open a clutch 41, in the opposite direction to the arrow 64, the control piston 56 contacts with its annular aurface 62 the annular surface 63 of the operating piston 54 and displaces the latter in the opposite direction to the arrow 64 until it contacts a housing projection 58 associated therewith. In this respect the bore 77 is closed by the operating piston 54 and the bore 78 is opened. The supply line 46 of the pressure control valve 40 is shut off and the fluid disposed therein is under the pressure P_{syst} permitted by the pressure limiting valve 45. The line 47 is connected, via the open bore 78, an annular duct 79 and via lines 59, 65 with the reservoir 43 and is therefore pressureless. The clutch 41 to be actuated is opened (point 0 in Fig. 1). If the clutch 41 is to be closed, the pedal 75 is released and the stem 74 follows the movement of the pedal in the direction of the arrow 64. The compression spring 55 displaces the operating piston 54 and the control piston 56 in the direction of the arrow 64 and the annular surface 63 of the operating piston 54 contracts the annular surface 62 of the control piston 56. The latter pushes against the stem 74.

The operating piston firstly closes the bore 78 and then opens the bore 77, so that the fluid flows from the supply line 46 to the line 47 and from there via the line 48 to the clutch 41 whose cylinder is filled. In the line 47 of the pressure control valve 40 there is therefore produced a predetermined filling pressure P_F in accordance in each case with the pressure set at the pressure limiting valve 45 as a function of the delivery of the pump 44, the flow resistances in the lines between the pump 44 and the clutch 41 and the mechanical resistances of the clutch piston. The compression spring 57 is not strong enough alone to press the operating piston 54 against the force of the compression spring 55 in the opposite direction to the arrow 64 (and to separate the annular surfaces 62, 63), to close the bore 77 and as result of this to throttle or discontinue the flow of the fluid from the supply line 46 to the line 47. This is only possible if the pressure in the line 47 has been transmitted via the control lines 49, 71 and a control groove 73 and a line 72 to the differential surface ΔA on the operating piston 54 and supports the compression spring 57. However this path is firstly blocked for the fluid by the retarding piston 52, the line 72 and the annular groove 73 are connected with a chamber 70 and therefore pressureless, as the chamber 70 is constantly in connection with the reservoir 43.

Whilst fluid is supplied to the clutch 41, a portion of the fluid flows via the supply line 49 and via the bore 51 in the diaphragm 50 and causes a displacement of the retarding piston 52 in the direction of the arrow 64 as a result of which the compression spring 53 is compressed. The deflection of the retarding piston 52 lasts until the retarding piston firstly closes the control lines 71, 72 and the control groove 73 and subsequently releases a passage from the control line 49 to the control lines 71, 72 and to the control groove 73. In this respect the fluid may press against the differential surface ΔA and together with the compression spring 57 displace the operating piston 54, as a result of which the free cross-section of the bore 77 and therefore the fluid flow are reduced such that the pressure prevailing in the line 47 and acting on the differential surface ΔA drops until the sum of the compressive force on the differential surface ΔA and the force of the spring 57 is equal to the force of the compression spring 55. The pressure at the valve outlet in the line 48 therefore decreases from point 3 (Fig. 1) to point 4.

If the pressure prevailing in the line 47 and acting on the differential surface ΔA is smaller than described, the force of the compression spring 55 is therefore predominant such that the operating piston 54 is again displaced in the direction of the arrow 64 and the fluid flow and therefore the pressure in the line 47 and acting on the differential surface ΔA again increase. However if this pressure is greater than described, the sum of the compressive force on the differential surface ΔA and the force of the compression spring 57 is predominant such that the operating piston 54 is displaced in the direction opposite to that of the arrow 64 and the fluid flow and therefore the pressure in the line 47 and on the differential surface ΔA are again smaller. The operating piston 54 therefore admits the exact amount of fluid to the clutch 41 as is required to maintain the required pressure, since the control lines 49 and 71 are connected together via the retarding piston 52.

The pressure flow coming from the pump 44 is discontinued to the differential surface ΔA until the retarding piston 52 has been displaced, under compression of the compression spring as a result of the pressure building up in the control line 49, to such an extent in the direction of the arrow 64 that it releases the opening to the differential surface ΔA. The duration t_v of the delay is determined in this case by the filling time of the retarding cylinder containing the retarding piston 52.

The diameter of the bore 51 in the diaphragm 50 is dimensioned such that the pressure in the line 48 decreases from the value P_F to a value corresponding to point 4 in the diaphragm of Fig. 1, shortly before the clutch 41 is completely closed.

Whilst the cylinder in which the retarding piston 52 extends is filled via the diaphragm 50, the pedal 75 may be further released in the direction of the arrow 64. The operating piston 54 and the control piston 56 are in this respect displaced in the direction of the arrow 64 until the operating piston 54 contacts the housing of the pressure control valve 40. Further movement of the pedal leads to the control piston 56

only continuing to be pressed by the compression spring 57 against the stem 74 and also displaced in the direction of the arrow 64, wherein the annular surfaces 62, 63 are separated from one another. As soon as the cylinder with the retarding piston is filled, the pressure acting on the differential surface ΔA causes a displacement of the operating piston 54 in the opposite direction to the arrow 74 until the flow of fluid through the bore 77 has been throttled to the required extent. The further the control piston 56 moves in the direction of the arrow 64, the weaker the force of the compression spring 57 is. The pressure acting on the differential surface ΔA must therefore be correspondingly higher in order to displace the operating piston 54 in the opposite direction to the arrow 64 under compression of the compression spring 55. The regulated pressure is therefore dependent on the positioning of the pedal 75.

If the pressure in the line 47 increases, corresponding to the respective position of the pedal 75, the pressure acting on the differential surface ΔA effects a displacement of the operating piston 54 in the opposite direction to the arrow 64 to an extent such that the bore 77 is closed and the bore 78 is partly opened and an exact amount of fluid escapes via the line 59 until the pressure in the line 47 has again reached the required level.

If the control piston 56 is displaced further in the direction of the arrow 64, it finally reaches a position in which it locks the control line 71 and causes the pressure on the differential surface ΔA to escape via the control groove 73 and the chamber 70 into the reservoir 43. The compression spring 55 then displaces the operating piston 54 in the direction of the arrow 64 up to the projection. The supply line 46 is directly connected with the outlet of the pressure control valve 40 via the bore 77, i.e. with the line 47. The pressure then increases from point 5 to point 6 in the diagram of Fig. 1 which corresponds to the system pressure P_{syst}.

If the pedal 75 is moved in the direction opposite to that of the arrow 64, the control function is again operative after connection of the control line 71 with the line 72 via the control groove 73, corresponding to the section to the section 5 - 4 in the curve of Fig. 1. In the case of increasing compression of the compression spring 57 the regulated pressure decreases in the line 47. When the control piston 56 contacts the operating piston 54 and displaces the latter in the opposite direction to the arrow 64 until the line 47 is connected with the line 59 and the line 65, the line 47 is pressureless. In a similar manner the control line 49 is pressureless and the retarding piston 52 displaces the fluid from the retarding cylinder. The diaphragm 50 thereby releases a greater cross-section so that the retarding cylinder is rapidly emptied.

The time delay time t_v may also be determined by other timing elements. In this respect use may be made, for example, of a timing element 95 (Fig. 6) which actuates a switch 97 for closing a circuit and which also comprises a battery 96 and the winding of a junction valve 94. In this embodiment a clutch 90 is to be actuated by a pressure medium. The latter is supplied from a reservoir 100 via a check valve 99 by a pump 98 which is disposed in parallel with a pressure limiting valve 107. A return line 101 leading back to the reservoir 100 is connected to a pressure reducing valve 93. A further junction valve 91 which is mechanically actuated and connected to a further return line 102 is mechanically linked to the pressure reducing valve 93. A further return line 103 leads from the first junction valve 94 back to the reservoir 100. A line 105 is also included for the mechanical connection between the pressure reducing valve 93 and the junction valve 91. The pressure reducing valve 93 and the junction valve 91 together form a pressure control valve with a compression spring 92. A line 104 forms a pressure connection between the pressure control valve 91, 93 and the junction valve 94.

The regulation of the pressure during the filling stage may also be delayed as a function of the piston stroke instead of a timing element. In this case there is no need for the timing element of the embodiment of Fig. 6. The switch 97 is, in this case, actuated directly or indirectly by the piston of the fluid controlled element when the empty stroke (or the major portion of the latter) is carried out. A hydraulic switch may also be envisaged at this location.

CLAIMS

1. A device for use in a fluid controlled system for rapid actuation of a pressure controlled cylinder, preferably acting as part of a clutch or brake, and having a pressure control valve connected in series to the cylinder, which valve is associated with a control line provided with a retarding device, said valve providing a through connection in its rest position and comprising a locking and throttling device for the main flow which has at least one control surface on a piston which bears against at least one spring, wherein the control line is part of a pressure medium actuated reversing device which comprises at least one return line.

2. A device as claimed in claim 1, wherein the pressure control valve is formed by an operating piston and a control piston.

3. A device as claimed in claim 1 or claim 2, wherein the retarding device is formed as a check diaphragm.

4. A device as claimed in claim 1 or claim 2, wherein the retarding device further comprises a retarding piston.

5. A device for use in a fluid controlled system for rapid actuation of a cylinder, substantially as hereinbefore described with reference to the accompanying drawings.

Printed for Her Majesty's Stationery Office by The Tweeddale Press Ltd., Berwick-upon-Tweed, 1983.
Published at the Patent Office, 25 Southampton Buildings, London, WC2A 1AY, from which copies may be obtained.